**RIVAS**

**Railway Induced Vibration Abatement Solutions**

**Collaborative project**

# Final report on the reference track design for vibration characterization of vehicles including draft measurement guideline for vehicle homologation

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# 1. EXECUTIVE SUMMARY

Railway vibrations are generated by the interaction between vehicle and track, and transferred through the ground to residential areas nearby. A system approach, addressing both the vehicle and the track, is needed to control these vibrations. Track related control measures are usually costly and carry the risk to be inefficient in cases where the vehicle performance is dominant. Vehicle properties to be controlled are the geometry (interference between track geometry such as sleeper distance and vehicle geometry such as axle distance and bogie distance), the wheel OOR and the unsprung mass, which is dominated to a certain extent by the performance of the bogie or axle suspension. For in service vehicles, wheel OOR can be detected by means of monitoring stations and can be cured by an optimised reprofiling regime. Some modification of current monitoring stations may be required, e.g. in the wavelength range detected by the system. Such monitoring stations may even be capable to detect the malfunctioning of the suspension system which is a further source for vibration-induced problems from in-service vehicles.

For newly designed vehicles, the optimal design depends on the performance of the track(s) which the vehicle will be operated on as well as on the soil properties. With the same vehicle, good system performance may be observed on one track and soil type, but may be significantly worse on another track and soil type. This makes it difficult if not impossible to test the vehicle performance on a real track. Strict requirements would have to be set to the test track, which would make it practically unfeasible to test the vehicle in practice. In task 1.4 of RIVAS, two methods have been developed which allow testing of the vehicle, either at standstill in a workshop or at low speed on a short stretch of track, assessing the vehicle receptance as the key indicator for the vehicle performance. Once this vehicle receptance has been assessed, the overall performance of the vehicle and track system can be predicted using the following parameters: track and soil receptance, combined unevenness of wheel and track, and soil transfer function. For a specific site, these parameters can be assessed with sufficient accuracy using the methods developed and/or defined in the RIVAS project. Alternatively, a standard track and soil type could be defined as a reference for a vehicle to be designed for operation throughout Europe. Such a standard track and soil need not to be real; virtual testing against such these standard track and soil properties could be the way forward to develop vehicles with better vibration performance.

The two methods developed in the current task are:

- Assessment of the vehicle receptance in the workshop, exciting the wheel with a shaker and measuring both the force and the displacement of the wheel,
- Assessment of the vehicle receptance on a stretch of track with a fabricated defect, e.g. a half sine shaped defect, which excites the vehicle when it rolls over the defect.

A simulation model of the track defect has been used to check whether a fairly limited and therefore realistic defect would generate enough vibrational energy into the wheel and track to be able to assess the vehicle receptance with sufficient accuracy. It turns out that this is a feasible method.

It is recommended that vehicle suppliers implement the latter method to gain further experience. In due course, the method could be developed into a standard method for vehicle testing.

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# 3. INTRODUCTION

## 3.1 General Aspects

The generation of railway induced ground vibrations originates from the interaction between vehicle and track. A system approach is required to control the generating mechanisms, roughly to be distinguished between quasi-static excitation and dynamic (including the so-called parametric) excitation.

The quasi-static excitation originates from the moving static load of the train, causing a force onto the track and soil beneath it, and travelling with the train speed.

The dynamic excitation originates from

- spatial variations in track geometry. These are generally indicated as unevennesses,
- spatial variations in track stiffness, usually called parametric excitations, causing an excitation with a distinct, characteristic frequency, e.g. the sleeper passing frequency,
- impact excitation from rail joints and wheel flats, with a broad band spectrum response,
- wheel irregularities. These are indicated as Wheel Out Of Roundness (Wheel OOR). Wheel OOR is distinguished according to the circular mode number:
    1. Eccentric wheels and oval wheels;
    2. Wheels with flats;
    3. Wheels with several irregularities with a distinct wavelength (called polygonisation).

These various types of dynamic excitation cause the wheel or axle and the track to be excited to vibrations. These vibrations are then transferred into the soil underneath the track and to possible residents nearby. To a certain part the generated vibrations are also transferred into the vehicle. The quality of the wheel or axle suspension determines the transfer of vibrational energy into the vehicle. For an optimal suspension, there is hardly any energy transferred into the vehicle for the frequency range relevant to ground vibrations. If this is the case, it is only the so-called unsprung mass of the wheel/axle that is excited to vibrate.

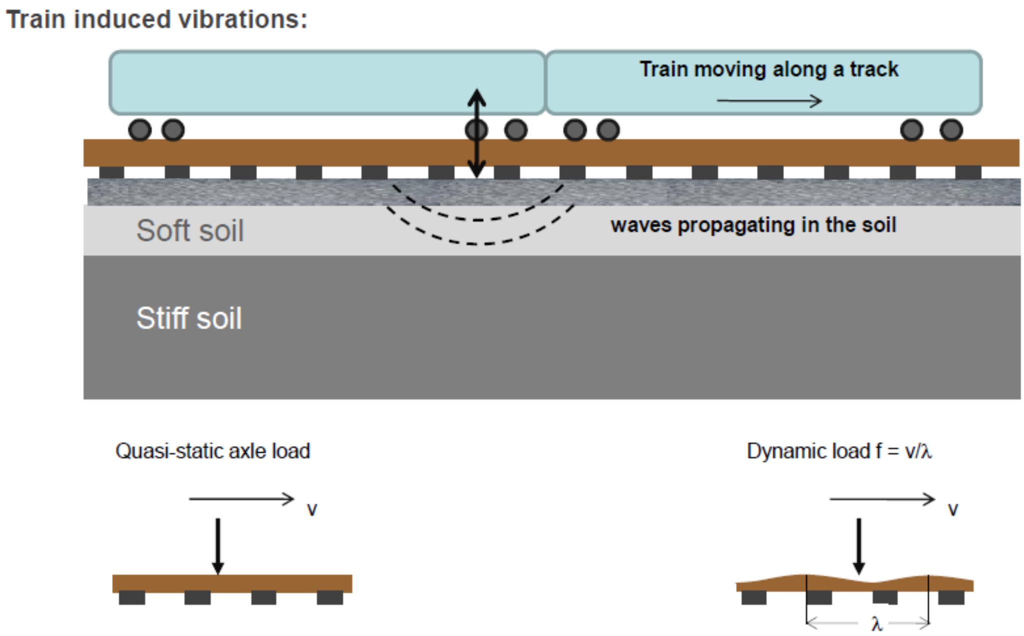


Figure 1. Two types of excitation from a moving train (from Bernd Asmussen,DB)

When considering mitigation measures, even within the system approach, the question arises whether one should start at the vehicle or at the track. An almost perfect vehicle, in terms of its performance with respect to its capability to generate vibrations, may still cause problems when the track properties cause high vibration levels. Vice versa, a well maintained and perfectly designed track may still cause high vibration levels in the ground when the vehicle is performing poorly.

Decisive for the optimal approach is on the one hand to find the "weakest link" in the system and on the other hand to identify the component optimisation with the highest impact. A full understanding of the whole system however is essential.

The vehicle designer will have to know most of the track parameters and the track and soil properties in order to design the vehicle such that the combination of vehicle and track and soil performs well and according to the specified expectation. This is a challenge since a new vehicle is usually not designed for just one particular track type, but should instead perform well on a wide range of different tracks.

And vice versa, it would be useless to try and design a perfect track – in terms of vibration performance – when the parameters of the vehicle which will run on that track are not known yet.

Work within RIVAS, in particular WP5, has however clearly demonstrated that there are significant opportunities to control vibrations by optimisation of the vehicle parameters. The main parameters to control in this respect are the vehicle geometry (axle distance in relation to sleeper distance, bogie distance), unsprung mass and wheel OOR. In the current report, we concentrate on the latter two parameters. The following table is quoted from [2], Guidelines for the design of vehicles, showing considerable reduction potential for ground borne vibrations by controlling the two main vehicle parameters, i.e. unsprung mass and wheel OOR. Based on this promising potential, the work in WP1.4 focuses on optimisation of these vehicle parameters, particularly on the way to predict, test and demonstrate the potential efficiency of such optimisation.

| | Feasible reduction of unsprung mass | Vibration reduction due to lower unsprung mass | Possible vibration reduction by reduction of OOR |
|---|---|---|---|
| Trailer bogies | 15 % | < 1 dB | < 5 dB |
| Powered bogie | 25 % | 1 – 2 dB | < 5 dB |
| Locomotives | 35 % | 2 – 4 dB | < 20 dB |

**Table 1. Vibration reduction potential by vehicle optimisation (from [2])**

Although the opportunities for vehicle optimisation are promising, it should be noted however that the efficiency of reduction of wheel OOR is limited to vibration frequencies from about 25 Hz upward [1]. In this frequency range the problems are often related to reradiated (groundborne) noise rather than feelable vibrations. Although reduction of the unsprung mass may be efficient at lower frequencies, in the range of excitation below about 30 Hz the main excitation comes from track irregularities. As a consequence, vehicle related measures may have good efficiency at low frequencies only if the track quality is good.

## 3.2 OBJECTIVES

The objective of Task 1.4 is to define the conditions for predicting, designing and testing new rolling stock with respect to its performance to generate vibrations in the ground adjacent to the track. In addition, the work in Task 1.4 may also serve to define maintenance regimes that will keep a train in an optimized condition with respect to vibration generation, even though it is in operation. In the Description of Work of the RIVAS project, task 1.4 was defined as follows:

> *This task shall focus on the interface between the vehicle and the track and shall produce a novel approach to separating the responsibilities for optimized vibration performance between vehicle manufacturer and railway undertaker on the one side and infrastructure manager on the other. It is expected that the results of this task may represent a basis for further work into an international technical specification, defining the constraints for vehicle and track at component level, but aiming at an improved performance at system level. The work will be coordinated with WP 2 [mitigation measures of vibrations at the source- pdv] and task 5.2[Characterization of vehicle parameters – pdv]. The results of task 1.4 work will feed into task 1.2 [Assessment of mitigation measures performance – pdv] where standard measurement procedures will be defined, and into task 2.4 [Solutions for optimised rolling stock maintenance - pdv] and Task 5.4 [Field test execution and data processing pdv]. A standard track quality in terms of input impedance[receptance – pdv] shall be defined, which is to be used as design parameter for the vehicle manufacturer, much like it was done in the current TSI Noise with the reference track roughness and decay rate for noise.*

As was pointed out in various documents within the RIVAS project, the generation of vibrations and ground borne noise can be described as the result of three subsystems, i.e. the vehicle, the track and the soil, and two interfaces, i.e. the wheel rail interaction and the track-subsoil interaction. Depending on the chosen approach, the building may represent a fourth subsystem, with the interface between soil and building foundation as a third interface.

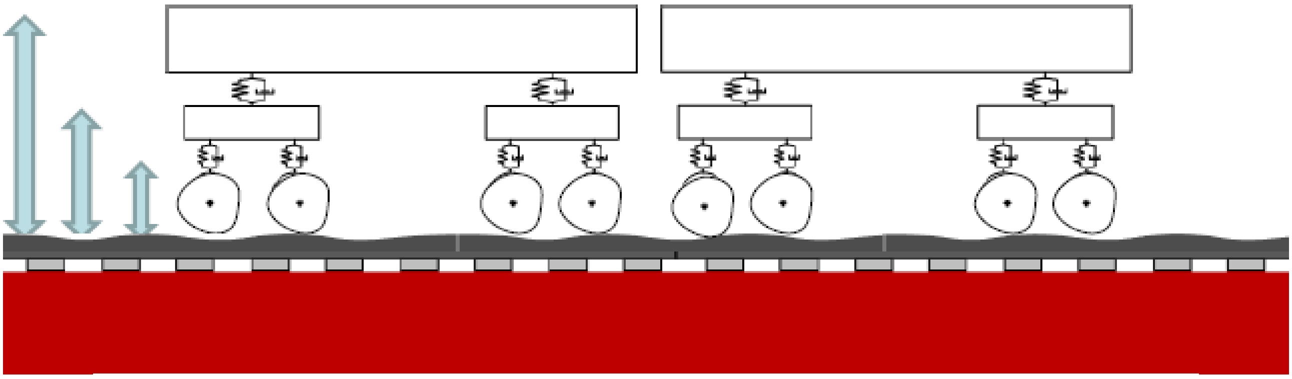

**Figure 2 . the vehicle model showing track unevenness, wheel out of roundness and primary and secondary suspension (from A. Mirza, Rivas 2nd workshop) with masses of the wheel, the bogie and the whole car**

The system approach referred to above is only possible when the parties responsible for the various subsystems collaborate. A vehicle supplier, designing a new vehicle for optimum vibration performance, can only do so when he is aware of the track and soil properties encountered by the vehicle in operation. And when it comes to testing the performance of a new vehicle, this can only be done on a well-defined test track with known properties, and – if the test refers to a certain maximum vibration level at some distance from the track – with a well-defined soil with known properties.

Originally, task 1.4 intends to define a particular standard track and a particular standard soil, in such a way that their influence in the total system would be minimized, aiming at a result where the vehicle influence would be maximized. If such a system could be created where the vehicle influence is maximal, testing a vehicle in practice within this system would clearly reveal weaknesses in the vehicle's performance. Such would for example be the case if the track were very heavy, very stiff and very even (low unevenness).

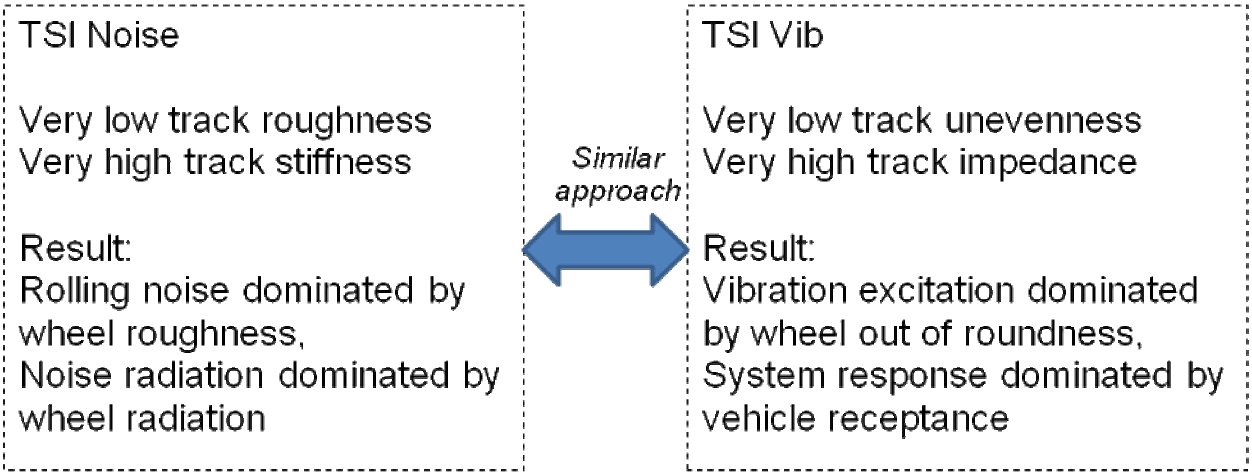


**Figure 3. Similar approach for vibration performance and noise performance of a railway vehicle**

Such an approach would correspond to the approach chosen for testing and designing vehicles with respect to their performance in generating rolling noise. The current testing method as defined in the TSI-Noise [3] assumes a track with very smooth rail head surface (in other words: low rail roughness) and high track decay rate (high stiffness). These requirements have been laid down in ISO 3095 [4]. This method for noise assessment has been in force since 2005; current experience shows that it is difficult to find sufficient tracks that comply with the ISO 3095 requirements [5]. Therefore a FP7 research project called Acoutrain aims at developing a method for so-called virtual testing of the vehicle, including a complete rolling noise model. Building on this experience, it was proposed in task 1.4 to consider a method of virtual testing of railway vehicles for their vibration performance. The current report describes both a method of practical testing and a method of virtual testing.

When considering a system of virtual testing for vibrations, it is helpful to realise that there is a big difference between the areas of noise and of vibration when it comes to the availability of reliable prediction and modelling methods. In several reports of the RIVAS project is has been made clear that it is hardly feasible to predict railway induced vibrations in adjacent buildings with the level of accuracy required for compliance testing. At least that is the case when the prediction would be based entirely on numerical models. The main reason for this lack of availability is the inherent fact that essential parameters vary significantly from one site to another; therefore some empirical assessment of such parameters will always be required as part of the prediction method. The current report identifies these parameters and proposes methods for their assessment for the application of vehicle testing. The term "virtual testing" will still be applicable to the methods described, as these will be a combination of measurement and computation.

## 3.3 VIBRATION PERFORMANCE OF A VEHICLE

### 3.3.1 Design process of a new vehicle

If a future train operator or train owner wishes to specify a certain vibration performance for the vehicle which is being ordered, the specification would generally be in terms of a maximum vibration velocity in a certain frequency range, possibly in three directions, at a certain distance from the track. In addition the client could provide the train designer with the key parameters of the track. These parameters would be:

- Rail type, rail mass
- Track type, ballasted or non-ballasted including ballast thickness
- Details on rail fasteners, including dynamic stiffness under load
- Possibly track decay rate

Note: the train operator or vehicle owner may have a problem to specify these properties; indeed, many railway vehicles are operated on many different tracks with quite different properties. In that case it would be recommended to first identify critical sites along the track, i.e. sites where sensitive buildings are located particularly close to the track. Such sites could be hot spots in terms of track maintenance and track renovation, as a first step to avoid problems with ground borne vibrations. Thus, a dialogue will start between the vehicle operator and the track maintenance manager, on the existing and required quality of the track. Once an agreement has been reached between these parties, the properties of that "minimum quality" track can be transferred to the vehicle designer.

Summarizing, we assume that the client (vehicle owner) is able to provide the vehicle designer with details on the track receptance of the track which the vehicle will mainly be operated on.

Note: the receptance is the amount to which an object displaces when a unit force is applied to it (displacement per unit force). A track with a low receptance shows a small displacement when the moving train exercises a force onto it.

Similarly, the vehicle designer should then work on the vehicle properties, expressed in terms of the vehicle receptance, a quantity determining the amount to which the vehicle will be displaced when a force is applied to it.

As an illustration, an example of the dependence of vehicle receptance from frequency for different values of the primary suspension stiffness is presented in the following figure (from A. Mirza, 3$^{rd}$ RIVAS workshop).

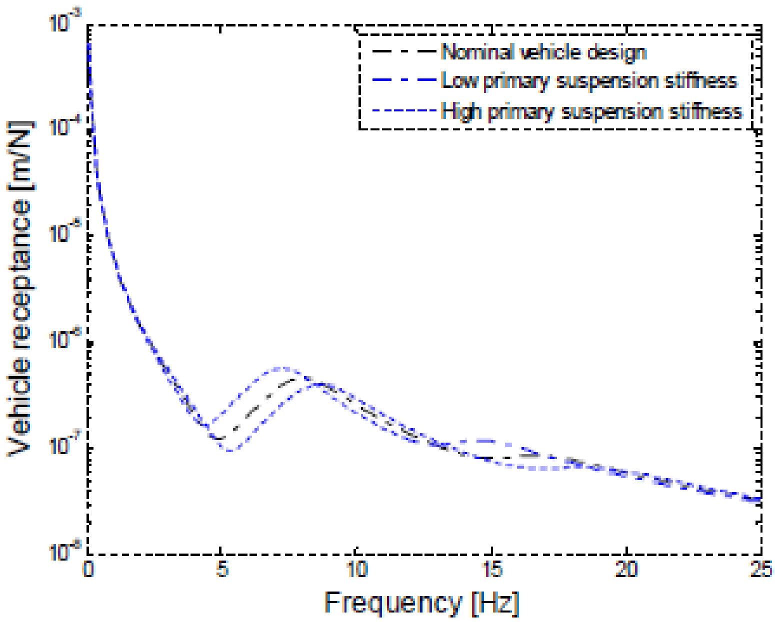


**Figure 4. Example of vehicle receptance as a function of frequency for three different primary suspension stiffnesses (from A. Mirza, 3rd Rivas workshop)**

The designer usually works with an expected vehicle receptance, based on measured receptance from vehicle types that have been developed before and that show analogy with the vehicle to be developed. Thus, the designer would perform a first check whether the current design is expected to comply with the client's specifications. In order to carry out that compliance check the vehicle designer would use a numerical prediction method, which has to be agreed with the customer. This requires particular agreement on the soil parameters to be used in the prediction model. This may require acquisition of typical soil parameters at a number of characteristic locations along the track which the vehicle is to be operated on. If compliance cannot be demonstrated, the vehicle design could be modified and improved in that stage of the design process. As soon as a first prototype is available, the vehicle manufacturer would check the real vehicle receptance and use the measured value as a parameter in the model. This would make the prediction more reliable. If the vehicle does not comply with the required specifications, modest modifications could still be introduced to the vehicle design. Alternatively, the client has to be informed and the specification revised.

In this process, an accurate assessment of the vehicle receptance is essential. Even if spatial variations in the soil properties cannot be predicted with great accuracy, the prediction model should be as accurate as possible on a limited number of well-defined sites along the track. These are the sites where soil data has been collected.

The above process refers to new passenger trains (multiple units, coaches and locomotives) for both conventional speed and high speed. The emphasis is on heavy rail although light rail is not excluded.

For freight cars the focus is more on the operational condition and the maintenance necessary to maintain a minimum quality, rather than on the performance of the vehicle when it is new. The maintenance of vehicles in operation is treated in the next section.

### 3.3.2 Maintenance of in service vehicles

For in service vehicles, both passenger and freight, the main characteristics decisive for the vehicle's vibration generating performance are wheel out of roundness (wheel OOR) and malfunctioning of the primary and secondary suspension (where applicable) of the wheel within the bogie.

Passenger train wheels are generally checked almost every 2 to 3 weeks for wheel OOR, mainly wheel flats and polygonisation. If the wheel unevenness exceeds certain limits, wheels will be reprofiled. Measurements in Switzerland show [D5.3], that wheels which are regularly checked and curatively reprofiled generally have similar quality with respect to vibration generation as e.g. freight wheels, which are checked only once in about. 6 years. The conclusion is that the current monitoring and reprofiling procedure is not adequate for vibration control. Most likely this is due to the low frequency wheel OOR relevant for vibration generation as opposed to the higher frequency wheel OOR which is covered by most monitoring stations.

Various networks and infra managers have developed and installed monitoring stations along the track, where the dynamic load to the track is measured. So, the monitoring is mainly focussed on the detection of wheel flats. The measurement is usually carried out by means of strain gauges or accelerometers on the rail. In combination with some form of automated vehicle recognition, these monitoring stations are capable to detect wheel flats on specific wheels. In WP5 of RIVAS, it was demonstrated (3rd RIVAS workshop, presentation Pau Gratacos) that improving the reprofiling procedure, in particular by removing less material more frequently instead of removing more material at longer intervals, would result in a considerably better wheel quality with respect to wheel OOR.

For in service vehicles, malfunctioning of the wheel suspension is probably more difficult to detect with sufficient certainty. There are indications that this occurs regularly with freight vehicles, due to the primary suspension (often flat springs) being contaminated or corroded, leading to the suspension being blocked i.e. short-circuited.

### 3.3.3 Vehicle's influence on ground vibrations

The excitation of ground-borne vibration from trains originates from the interaction between the vehicle and the track.

The broadband dynamic excitation caused by wheel and rail irregularities (wheel OOR and track unevenness) will excite resonances in the vehicle-track system. The vehicle itself has primarily two resonances important for ground vibration: the car body bouncing on the secondary suspension and the bogie (or axle) bouncing on the primary suspension. Furthermore the unsprung mass of the wheel or bogie will bounce on the track stiffness at a particular frequency which commonly is referred to as the P2 resonance. On a passenger vehicle the car body bounce is often seen below a few Hz while the bogie bounce is found in the 8-15 Hz range. The P2 resonance which is determined by the weight of the unsprung mass and the track stiffness will vary from one track to the other and is hence not a characteristic of the vehicle only. This resonance is usually located in the 80-100 Hz range. At these resonances the broadband dynamic excitation can lead to large amplitudes in the vibration response. Extensive simulation studies within RIVAS have shown that the influence of the car body bounce is often masked by the high level of quasi-static excitation at low frequencies. The bogie bounce resonance is seen as a narrow band influence in the vibration response which could be problematic if the ground response is high in that particular frequency range. The decoupling of the bogie mass from the unsprung mass will be shifted towards higher frequencies for a stiff primary suspension. This could lead to a stronger parametric excitation for train speeds where the sleeper passing frequency is lower than the bogie bounce resonance.

The weight of the unsprung mass will influence the emission of ground vibration in two ways. The size of the mass, in combination with the track stiffness, determines the P2 resonance as described above. Additionally, a large unsprung mass will lead to higher inertia forces when the wheel is negotiating the wheel and rail irregularities. The high inertia forces of a heavy unsprung mass result in high contact forces between wheel and rail which in turn excites more ground vibration.

Finally the distribution of wheel axles, which is determined by the distance between adjacent axles and adjacent bogies, may also influence the excitation of vibration. The repeated passages of individual axles will introduce a dynamic fluctuation to the excitation from quasi-static axle loads. The frequency of this excitation is determined by the distance between the axles and the speed of the vehicle. This geometric resonance can often be seen in the vibration response spectrum of the ground. Whether or not this form of narrow band excitation is problematic will depend on the vibration response of the particular site. The simulation and measurement studies within RIVAS have shown that this effect often is of less importance in comparison with the influence from e.g. the unsprung mass.

### 3.3.4 Recommendations to the vehicle designer

From the above we can derive a series of recommendations for the vehicle designer who intends to optimize the vehicle in terms of its ability to contribute to the generation of ground vibrations:

- Reduce the unsprung mass
    - for locomotives and power cars in multiple units: avoid or minimize installation of heavy weight machines, such as gearboxes and transformers in the bogie.
    - for locomotives and power cars in multiple units: improve the tuning of the secondary suspension inside the bogie so that the heavy equipment such as gearboxes etc. is suspended resiliently.
    - for trailing bogies: reduce the mass of the axle (studies of Lucchini have indicated a mass reduction potential of down to 70% of the original mass).
    - for trailing bogies and axles: reduce the mass of the wheel (studies have indicated a potential reduction of 5 to 10% of the wheel mass, by reducing the wheel tyre radial thickness, possibly at the expense of reduced lifespan of the tyre as well as potentially increased wheel noise in the audio frequency range)
- Maintain the wheel OOR at a possibly low level (from practical tests, it was found that there may be up to 40 dB difference in wheel OOR between “good” and “bad” wheels).
    - for in-service vehicles: check wheel OOR more frequently, tighten the threshold for reprofiling (the amount of material to be taken away in reprofiling can be reduced drastically, if one carries out the reprofiling more frequently, so the effect on wheel wear would be only marginal or is even likely to be positive)
    - for new vehicles: improve the braking system, particularly the anti-sliding system which influences the growth of wheel OOR significantly

- for new vehicles: improve the wheel material in order to make the wheel less prone to wheel OOR, and
- improve the bogie design in terms of tuning the suspension systems to the properties of the track where the vehicle will be operated

All of the above will improve the overall system performance mainly in the frequency range between 30 to 100 Hz. If one needs to improve lower frequencies, then one would have to concentrate on the track properties (10 – 50 Hz) and/or the soil (5 -100 Hz). For these frequency ranges, the vehicle properties can only partially be held responsible.

### 3.3.5 Summary: the relevance of a standard track

In the previous sections we have introduced the challenges of the vehicle designer and vehicle maintenance manager in improving the vibration performance of a railway vehicle.

- *In the design stage*, a standard track (and soil) needs to be agreed with the customer, so that the design of the vehicle can be done against the parameters of this standard track. It would probably perform best on a track that resembles the standard track.
- *During the design stage,* the designer would want to check whether or not the vehicle performs as expected on the agreed standard track. This track however is not always available in practice. Therefore, the designer needs to have a well defined reference track available, on which he can then test the actual prototype vehicle and – if necessary – further improve it.
- *The final prototype, i.e. the real vehicle,* should also be tested on this reference track, so that the essential vehicle parameters can be assessed and collected. This would allow to transfer these parameter values to any real track situation and to predict the performance of the vehicle under real conditions. This would represent the compliance test, modified as a virtual test, because it is only carried out on the well-defined reference track and not in reality. The reason why the test would not be carried out in reality is that the results would depend too much on the local situation (soil properties) and on the incidental track quality (track unevenness), which may change over time.

# 4. PROCEDURE FOR COMPLIANCE TESTING OF NEW VEHICLES

## 4.1 SELECTING THE TRACK AND GROUND PARAMETERS

It has been pointed out in the previous chapter, that the vehicle designer has good opportunities to improve the vibration performance of the vehicle, by carefully tuning the geometrical properties of the vehicle, by reducing the unsprung masses of the wheel and bogie, by carefully designing the suspension properties and by avoiding and removing possible wheel OOR. When doing so, the vehicle designer may want to have detailed information on the track and soil properties for the track where the vehicle will be operated. This section attempts to assess these properties.

Once the first prototype of the vehicle is available, both the designer and the operator may want to check the compliance of the vehicle against the specifications. And certainly when the final design is finished and the vehicle is about to be delivered, compliance tests would be needed to demonstrate the performance of the vehicle. To this effect, a test site would be needed with "standard" properties both for the soil and the track. It would be advantageous if both the soil and the track properties would be representative for the "real life" operation conditions of the vehicle. On the other hand, if standard specifications for vibration performance would be targeted, such as in a TSI-Vib, then both the soil and track properties would have to be representative for the 'standard' European situation. The following paragraph intends to derive standard soil properties typical for the European situation.

### 4.1.1 Ground parameters

In RIVAS WP 4, mitigation measures in the propagation path were considered. Because the soil plays a very important role for their mitigation effect, soil conditions for seven different European test sites were detected and described within the RIVAS project [6]:

*"The RIVAS test sites are the Gerona test site (Spain) and the Furet test site (Sweden). These sites are supplemented by the site at Lincent (Belgium) with alluvium soil deposits, the Groene Hart site (the Netherlands) with heavily stratified soil, the site at Horstwalde (Germany) with homogeneous soil conditions, the site at Steventon (United Kingdom) with a clayey soil, and the site at Ledsgård (Sweden) with an extremely soft organic soil layer. The locations of the proposed reference sites are indicated in figure 5. In this way, a wide range of soil conditions, representative for different regions in Europe, is considered in the parametric study of the mitigation measures in the transmission path. Furthermore, the geophysical features (homogeneous soil conditions, soft top layer,. . ) are shared by many other sites in Europe where problems with railway induced vibrations may occur. "*

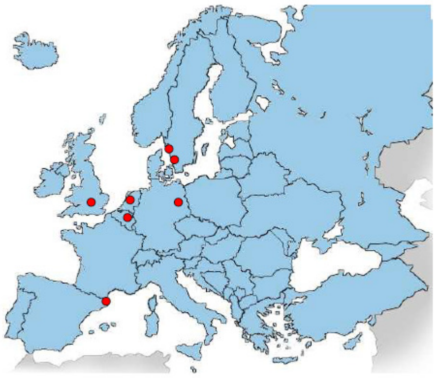

**Figure 5. Test locations for the RIVAS project**

To describe the soil conditions, mainly five parameters per layer are required. For both the shear and the dilatational wave, the wave velocities and the damping parameter are needed. Moreover the densities of the soil layers have to be known. Parameters can be determined by a combined measurement of the spectral analysis of surface waves and of the measurement of seismic refraction test [Del. 1.2 Annex]. In addition, the density is determined by laboratory investigation of a soil sample. The results are summarized in the following tables:

Horstwalde (Germany):

| Layer | $h$ [m] | $C_s$ [m/s] | $C_p$ [m/s] | $\beta_s$ [-] | $\beta_p$ [-] | $\rho$ [kg/m$^3$] |
|---|---|---|---|---|---|---|
| 1 | $\infty$ | 250 | 1470 | 0.025* | 0.025* | 1945 |

Lincent (Belgium):

| Layer | $h$ [m] | $C_s$ [m/s] | $C_p$ [m/s] | $\beta_s$ [-] | $\beta_p$ [-] | $\rho$ [kg/m$^3$] |
|---|---|---|---|---|---|---|
| 1 | 1.4 | 128 | 286 | 0.044 | 0.044 | 1800* |
| 2 | 2.7 | 176 | 286 | 0.038 | 0.038 | 1800* |
| 3 | $\infty$ | 355 | 1667 | 0.037 | 0.037 | 1800* |

Gerona (Spain):

| Layer | $h$ [m] | $C_s$ [m/s] | $C_p$ [m/s] | $\beta_s$ [-] | $\beta_p$ [-] | $\rho$ [kg/m$^3$] |
|---|---|---|---|---|---|---|
| 1 | 1.0 | 275 | 740 | 0.025* | 0.025* | 2000* |
| 2 | 1.0 | 325 | 740 | 0.025* | 0.025* | 2000* |
| 3 | 4.0 | 380 | 1450 | 0.025* | 0.025* | 2000* |
| 4 | 7.0 | 470 | 2280 | 0.025* | 0.025* | 2000* |
| 5 | $\infty$ | 600 | 2580 | 0.025* | 0.025* | 2000* |

Furet (Sweden):

| Layer | $h$ [m] | $C_s$ [m/s] | $C_p$ [m/s] | $\beta_s$ [-] | $\beta_p$ [-] | $\rho$ [kg/m$^3$] |
|---|---|---|---|---|---|---|
| 1 | 2 | 154 | 375* | 0.025 | 0.025 | 1800 |
| 2 | 10 | 119 | 290* | 0.025 | 0.025 | 1850 |
| 3 | $\infty$ | 200 | 490* | 0.025 | 0.025 | 1710 |

Groene Hart (Netherlands):

| Layer | $h$ [m] | $C_s$ [m/s] | $C_p$ [m/s] | $\beta_s$ [-] | $\beta_p$ [-] | $\rho$ [kg/m$^3$] |
|---|---|---|---|---|---|---|
| 1 | 3.7 | 50.0 | 1761.0 | 0.025* | 0.025* | 1107.1 |
| 2 | 7.0 | 75.0 | 1719.3 | 0.025* | 0.025* | 1500.0 |
| 3 | 8.3 | 180.0 | 1685.5 | 0.025* | 0.025* | 1970.6 |
| 4 | 9.3 | 240.0 | 1715.1 | 0.025* | 0.025* | 1970.6 |
| 5 | $\infty$ | 260.0 | 1726.1 | 0.025* | 0.025* | 1970.6 |

Steventon (Great Britain):

| Layer | $h$ [m] | $C_s$ [m/s] | $C_p$ [m/s] | $\beta_s$ [-] | $\beta_p$ [-] | $\rho$ [kg/m$^3$] |
|---|---|---|---|---|---|---|
| 1 | 0.7 | 120 | 1700 | 0.075 | 0.075 | 2000* |
| 2 | 2.0 | 200 | 1700* | 0.075 | 0.075 | 2000* |
| 3 | 3.0 | 120 | 1700* | 0.075 | 0.075 | 2000* |
| 4 | $\infty$ | 400* | 1700* | 0.075 | 0.075 | 2000* |

Ledsgard (Sweden):

| Layer | $h$ [m] | $C_s$ [m/s] | $C_p$ [m/s] | $\beta_s$ [-] | $\beta_p$ [-] | $\rho$ [kg/m$^3$] |
|---|---|---|---|---|---|---|
| 1 | 0.8 | 60 | 300 | 0.04 | 0.04 | 1700 |
| 2 | 3.2 | 44 | 570 | 0.04 | 0.04 | 1260 |
| 3 | 2 | 49 | 1050 | 0.04 | 0.04 | 1450 |
| 4 | 2 | 56 | 1050 | 0.04 | 0.04 | 1450 |
| 5 | 15 | 75 | 1050 | 0.04 | 0.04 | 1500 |
| 6 | 18 | 106 | 1050 | 0.04 | 0.04 | 1600 |
| 7 | $\infty$ | 132 | 1050 | 0.04 | 0.04 | 1700 |

The conclusion of this investigation is that the conditions in the selected sites show large variations. In summary, there is *no such thing as a "typical European soil"*. Because the vibration emission strongly depends on the soil parameters, calculations have to be performed for each test site by using the dynamic soil parameters as determined from measurements.

For vehicle design, the designer and the vehicle owner/operator would therefore have to agree on a range of different typical sites, each with their own soil properties, and the performance of the vehicle would have to be assessed against each of these properties.

### 4.1.2 Track parameters

For vehicle design, the designer and the vehicle owner/operator would have to agree on a range of different typical sites, each with their own track properties, and the performance of the vehicle would have to be assessed against each of these properties. If we intend to develop a standard European specification for vibration performance of vehicles, such as in a TSI-Vib, the track properties would have to be specified for a standard track, typical for the European situation.

Similar to the soil, there is wide variety of different tracks in Europe. Particularly the track unevenness and the track impedance may vary drastically in time, from one site to another and from one infrastructure manager to another.

For the purpose of compliance testing, we would like to have a track that reveals the weaknesses of the vehicle. This would have to be a track with very low unevenness (a very straight, smooth track) and with low receptance (high impedance, hence high track decay rate). Although such tracks may be found in practice, it is not practicable to define such a track as the standard measurement track for practical compliance testing. Therefore, the way forward would be to carry out measurements of the relevant parameters on a well-defined track and then transfer the results to the track and soil type relevant for the vehicle's operation. That is what is proposed in the next sections.

## 4.2 Motivation for virtual testing

As pointed out by Bongini et. al. [7], in the 7 years of TSI Noise being in force, some serious drawbacks have been identified. Firstly, the testing procedure is considered far too complex, too comprehensive and too costly in comparison with the minor changes of an existing vehicle design which sometimes cause a necessary certification. The same may apply to the degrees of freedom of the railway vehicle designer; in practice, there is fairly little variation between vehicles in terms of axle distance, wheel diameter and suspension type. In addition, with TSI Noise it was found difficult if not impossible to identify stretches of real track in full compliance with the requirements of TSI Noise, to gain access to that particular stretch of track, to transport the new vehicle to that site and to maintain these stretches in the appropriate condition. The same would probably apply to a uniform test track for vibration testing somewhere in Europe.

### 4.2.1 Excitation mechanisms: track unevenness, wheel OOR, parametric excitation

The excitation of ground-borne vibration, as previously described, is strongly influenced by the levels of wheel and track irregularities. Furthermore the parametric excitation caused by the variation in track stiffness introduced by discrete sleeper supports will also be of importance. When homologating a new vehicle according to the TSI Noise [3] a pre-described track roughness spectrum limit should be fulfilled by the test track. The same type of pre-defined excitation level could be used when simulating the ground vibration emissions caused by a new vehicle. Today, standard spectra of track irregularities are available for the application of railway noise in the TSI Noise and the ISO 3095 [4] standard and for the purpose of evaluating running stability and comfort, e.g. the ORE (ERRI) spectra. Noise is primarily excited by short wavelengths up to a few decimetres while comfort issues often are related to wavelengths of several meters. The relevant wavelength range for ground vibration is commonly located in the gap between these two ranges, i.e. from a few decimetres up to a few meters. No standard wheel or rail irregularity spectrum intended for ground vibration evaluation is available and hence such a spectrum would need to be developed by combining the data from short and long wavelength irregularity measurements. Figure 6 shows some examples of track irregularity spectra which have been used as input to the simulations done in RIVAS.

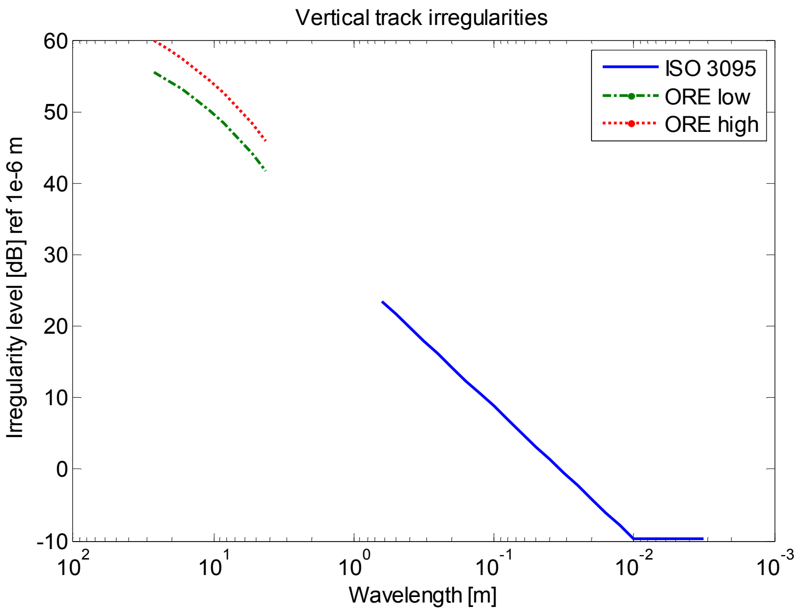


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**Figure 6 Vertical track irregularity spectra according to ISO 3095 [4] for noise tests and the ORE high and ORE low for vehicle dynamics**

Wheel irregularities and OOR will influence the excitation in the same way as the rail irregularities. Wheel OOR is especially important to consider for in-service vehicles since wheel damages or irregular wear on individual wheels may cause high excitation. For new vehicles however the wheels are expected to be in good shape and hence the excitation from wheel irregularities should be secondary compared to track irregularities and parametric excitation. Suitable input of wheel irregularities to the simulations could be prepared by studying measurement data from wheels on new vehicles.

A procedure for evaluating the performance of new vehicles in terms of ground vibration emissions could include both a number of standard spectra for vertical track irregularities which are representative for different situations or different parts of Europe. Additionally the option of using a site specific measured spectrum could be included for the customer and vehicle manufacturer to agree on. This would e.g. allow an operator of a metro network to put requirements on new metro vehicles based on the particular circumstances valid on the network where the vehicles will operate. The distance between sleepers may also vary among different networks and hence could also be adjusted in the simulations to better represent local conditions.

## 4.3 How to transfer measurement results to specific sites?

In the current chapter, it is investigated how the results from a particular, well-defined measurement situation could be transferred to any other, practical situation, in order to be able to predict the ground borne vibration response of the ground in the latter situation. In order to do so, we first investigate the parameters needed for such a transfer.

Assuming that the relative displacement excitation is the predominant source of excitation, the physical phenomena involved in the generation and propagation of ground vibration due to rolling stock pass-by can be expressed according to the following figure 7:

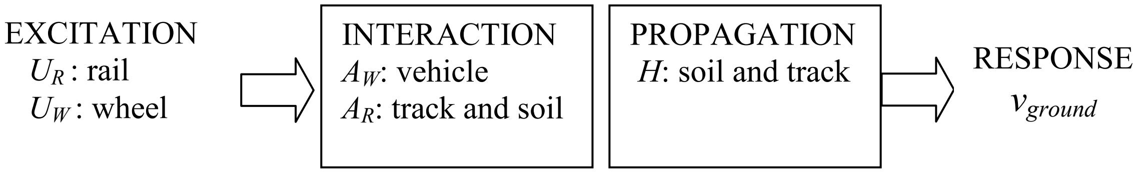


**Figure 7. Representation of the main parameters involved in the ground borne vibration excitation**

Thus, the response in ground vibration $v_{ground}$ can be calculated if we know the combined rail and wheel unevenness $U_c$, the receptances $A_W$ and $A_R$ of the vehicle and the track/soil-combination and the transfer function $H$.

If accurate predictions of the ground vibration are needed for a specific site, the parameters $U_R$, $A_R$ and $H$ have to be measured. Within the RIVAS project, measurement procedures for these parameters were described in detail [8]. If a direct measurement of one of the parameters is not possible, also indirect measurements (e. g. the stiffness of resilient elements and the mass of track components) can be carried out and the results coupled to simulations as shown in the following figure 8:

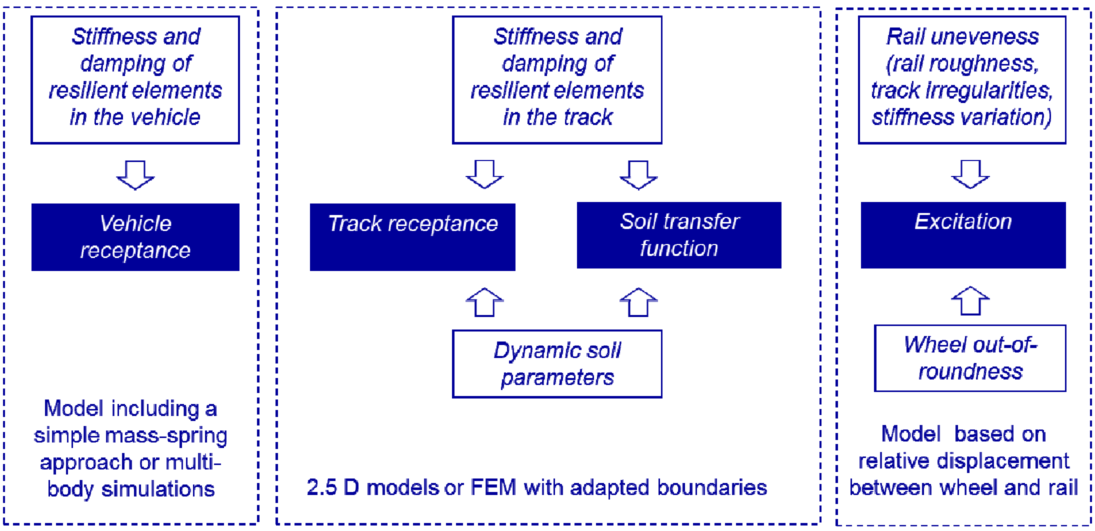


**Figure 8. Parameters in ground vibration prediction**

Although parametric excitation is not included in this approach, effects arising e. g. by stiffness variations along the track or sleeper passing effects might be included by using an “effective” rail unevenness. From figure 7 can be derived, that once the combi-ned unevenness $U_c$, the track and soil receptance $A_r$ and the soil transfer function $H$ are known, the outcome is determined by the vehicle receptance $Aw$. This means that the compliance test of a ve-hicle’s vibration performance can be reduced to a practical assessment of the vehicle receptance $A_w$.

## 4.4 PRACTICAL ASSESSMENT OF THE VEHICLE’S RECEPTANCE

### 4.4.1 Practical tests

Two types of measurement methods can be considered to measure the vertical receptance of the rolling stock seen from the rail:

- the **direct measurement** method on the stationary vehicle and
- the **indirect measurement** under rolling conditions.

Both methods are presented here and simulations illustrating the indirect method are presented.

### 4.4.2 Direct method on the stationary vehicle

The principle method consists of exciting the wheel directly with a shaker. Although this is fairly straightforward method, it has not been tested yet and was not proposed or described anywhere in literature for the application under concern here.

The train is in a workshop. All wheels are supported by the rail apart from the excited wheelset, which is supported, through its axle-boxes, by flexible springs.

The shaker is fixed under the wheel and the vehicle receptance is measured by recording the injected force at the shaker head and the wheel’s vertical response, measured with an accelerometer close to the excitation point.

**Advantages**:

- Direct and accurate measurement of rolling stock receptance seen from the rail.
- This measurement can be made in a workshop. It does not require rolling measurements.

**Drawbacks**

- This measurement requires a powerful excitation device. As a matter of fact, injected power should be sufficiently high to generate vibrational power the primary suspension significantly higher than frictional power. This could be checked by measuring the relative dynamic deflection of the primary suspension with a displacement sensor.

In practice, the force delivered by the shaker should be sufficiently high over the 0 – 10 Hz range, where the wheel set impedance is lower than the primary suspension one (above about 10 Hz, most of the injected force will be ‘absorbed’ by the motion of the wheel set).

***A minimum of 5,000 N, ideally 10,000 N should be considered.***

- For trains with different bogie types, a measurement should be made for each type of bogie.

### 4.4.3 Indirect method on the rolling vehicle

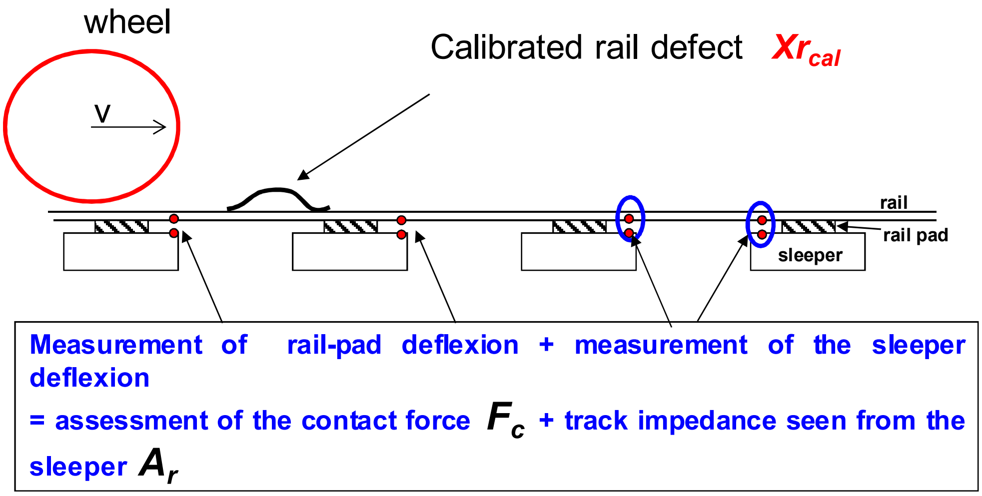


**Figure 9. Set up of the indirect vehicle receptance measurement method**

The above figure 9 represents the set-up for the indirect method to assess the vehicle receptance during pass-by of the vehicle. The method is based on excitation of the vehicle through and artificial rail defect, represented in figure 9 as a “hump”. The method is entirely new and was developed within RIVAS. The method has not been tested in practice yet.

The indirect method is based on equation (1):

$$F_{contact} = \frac{U_R + U_W}{A_R + A_W} \tag{1}$$

The method consists of rolling the vehicle over a calibrated rail defect: $U_{R\ Cal}$ of which the amplitude will be much higher than the wheel amplitude, on a track section with known receptance $A_R$ . The force injected between the rail and sleeper – named $F_{railpad}$ should be measured.

For stiff rail support, with a typical dynamic stiffness of 100 MN/m or more, one can demonstrate that below about 70 Hz, for the contact force $F_{contact} \approx F_{railpad}$ .

For higher frequencies, $F_{railpad}$ can be corrected by the mass effect of the rail, to obtain a more accurate value of $F_{contact}$ using the following formula:

$$F_{contact} = F_{railpad} + m.G, \qquad (1)'$$

*m* being the mass of the rail section above the rail-pads and ***G*** the average rail acceleration.

Equation (1) gives: $F_{railpad} \approx F_{contact} = \left(\frac{U_{R\ Cal}}{A_R + A_W}\right)$

so we have: $$A_W \approx \left(\frac{U_{R\ Cal}}{\mathrm{F}_{\mathrm{railpad}}}\right) - A_R \qquad (1)''$$

The vehicle vertical receptance seen from the wheel can be derived from formula (1)’’.

**Advantages:**

- The method can be applied on short test tracks that can be found on some train production plants (a few hundred meters track is sufficient since the method is supposed to be implemented for train speeds lower than 15 km/h).

- All axles on the test train are tested at the same time, which can provide information about receptance scattering between axles and will reveal the weaknesses in the design of the train (if any) more clearly thanks to them emerging repetitively.

- Measurements are performed in real rolling conditions and so non-linearities related to friction inside suspensions are accounted for.

**Drawbacks:**

- A calibrated defect has to be fixed on one the rails: a ‘half sine’ shape of about 5 to 10 cm long and of 1 mm height passed at a speed of about 15 kph will generate a dynamic force of about 50 000 N.

- The force transferred in the 4 rail supports surrounding the defect should be measured (see potential method for force measurement, next section). The easiest way to estimate this force is to mount linear rail-pads with a stiffness that has been measured in a laboratory (load versus displacement curve and dynamic stiffness).

- The track receptance at measurement section has to be be determined (see method proposed in annex).

- A preliminary calibration of the measuring system will be advised to improve the accuracy of the measurement, especially for the phase measurement which requires an accurate correspondence between wheel position and track force measurement. Ideally, this calibration should be carried out with a single rolling axle with known mass.

## 4.5 Illustration of indirect method with simulations

Simulations are presented in this section in order to propose different types of calibrated defects allowing to implement the indirect method and visualize the method using simulations.

### 4.5.1 Preliminary calculation

The rail defect is mainly defined by its shape $U_{R\,Cal}(x)$

$L$ is the defect length, and $Z_0$ the defect maximum height.

The length defines the frequency band of excitation [ 0 – $f_0$], with $f_0 = V/L$, where $V$ is the vehicle speed.

For a frequency $f_0$ of about 50 Hz and a typical speed $V$ of 3 m/s [appr. 11 kph] a defect length of $L$ = 0.06 meter should be considered.

For a sine shape $U_{R\,Cal}(x) = Z_0[\cos(2\pi \frac{x}{L}) + 1]$ from $x = -L/2$ and $x = +L/2$, circulated with a speed $V$, so with $x = V.t$,

and for low frequencies where $Ar \ll Aw$, $U_W \ll U_R$ and $Aw \approx 1/M.\omega^2$ with $M$ = semi-axle mass, equation (1) gives:

$$F(x) \approx U_{R\,Cal}(x)\ /Aw = U_{R\,Cal}(x)\ .\ M.\omega^2 \qquad (2)$$

so $$F(x) \approx Z_0[\cos(2\pi \frac{x}{L})]\ .\ M.\ 4.\pi^2.(V/L)^2 \qquad (3)$$

The maximum force value when reaching the top point at $x = 0$ is given by:

$$F = 8M.\pi^2.(V/L)^2.\ Z_0 \qquad (4)$$

***Numerical application***: for $M$ = 600 kg, $V$ = 3 m/s, $L$ = 6 cm and $Z_0$ = 0.5 mm

one has: $F \approx$ **60 000 N**

***Conclusion***: a typical rail defect of 1 mm amplitude peak to peak together with a rolling speed of 3 m/s and a defect length of about 6 cm should generate a sufficiently significant dynamic force amplitude of about 60 000 N.

### 4.5.2 Numerical simulations

In order to set-up a basis for rail defect definition and in order to illustrate the method, 4 different rail defects are simulated numerically:

- A rail joint of 6 cm length
- A rail joint together with a step down of 3 cm length and 1 mm depth
- A sine defect of 1 mm height peak to peak with a length of 6 cm
- A triangle defect with a 6 cm length and 1 mm amplitude (easier to manufacture)

The four defects are visualized in figure 10.

They all generate a peak to peak wheel/rail relative displacement of 1 mm (see figure 11).

The simulations are carried out with the GROUNVIB software developed by Vibratec. A 3 m/s train speed (10.8 kph) is considered for all simulations. The main input parameters are the following:

***Rolling stock parameters***

Axle mass per wheel: 600 kg

Bogie mass per wheel = 1 900 kg

Coach mass per wheel = 7300 kg

Primary suspension per wheel: stiffness = 7.5 MN/m damping = 11 400 Ns/m

Secondary suspension per wheel: stiffness = 0.4 MN/m damping = 8 200 Ns/m

***Track parameters***

UIC 60 rail: - rail-pad stiffness per rail = 217 MN/m$^2$

- rail-pad damping loss factor: 0.25

- Sleeper mass = 270 kg/m

Ballast and soil stiffness = 217 MN/m$^2$

Ballast and soil damping loss factor: 0.5

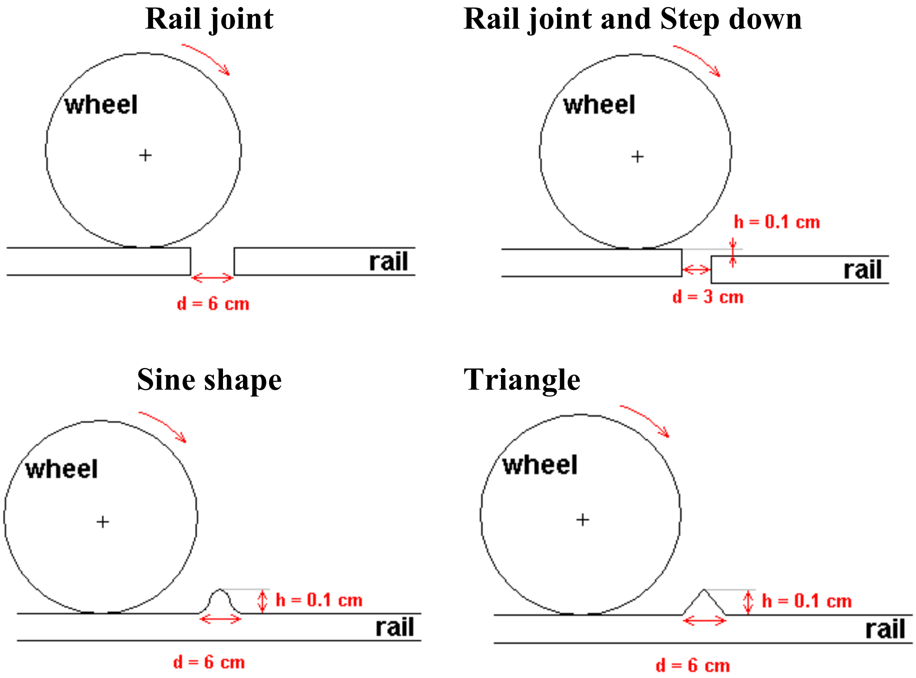


**Figure 10 – four different simulated rail defects**

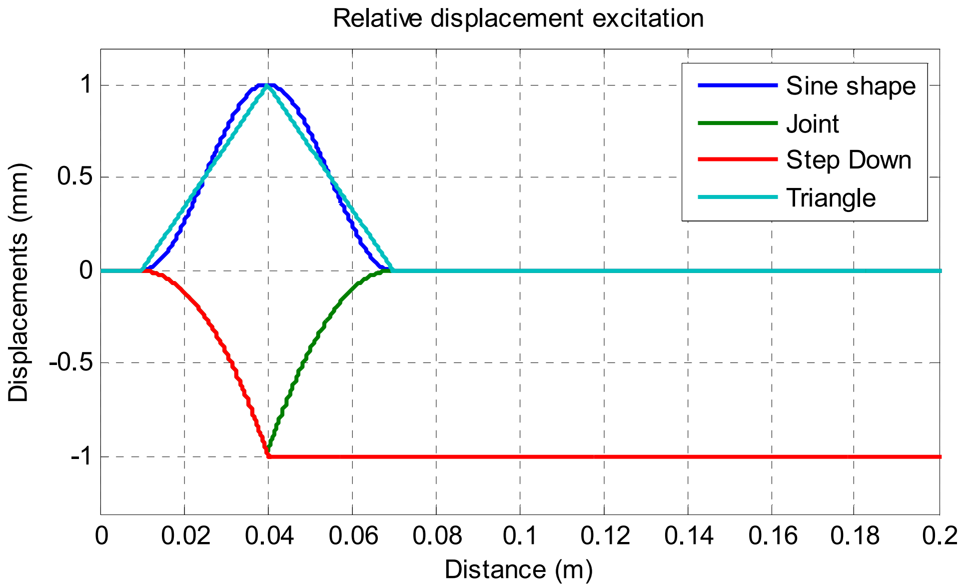


**Figure 11 – relative wheel/rail displacement induced by the 4 simulated defects**

Dynamic amplitudes generated by all defects are shown in figure 12; corresponding spectra are given in figure 13.

Additional dynamic forces related to the defect pass-by at 3 m/s, are ranging from 30 000 to 60 000 N according to the type of defect (peak value); these values are sufficiently high to generate vibrations on the measurement section higher than vibration related to wheel and rail unevenness (wheel unevenness of the test train should be low with no flats – rail unevenness should also be reasonably low).

- The sine shape defect generates a dynamic force of about 50 000 N peak amplitude which is close to the 60 000 N estimated value.
- The dynamic force generated by the triangle defect is close to that of the sine, but with slightly lower value (about 40 000 N instead of 50 000 N). Force spectra related to sine and triangle defects are close.
- The force generated by the rail joint of 6 cm length is quite high (60 000 N peak value); consequently a joint of 5 cm should be sufficient.
- The step down rail joint generates a dynamic force of about 30 000 N amplitude.

For all defects, the dynamic force generated by the defect becomes negligible about 40 cm away from the defect. Consequently, the dynamic force in the track should be measured on 4 adjacent sleepers only, from 1 sleeper before the defect to 2 sleepers after the defect, the defect being located just above a sleeper.

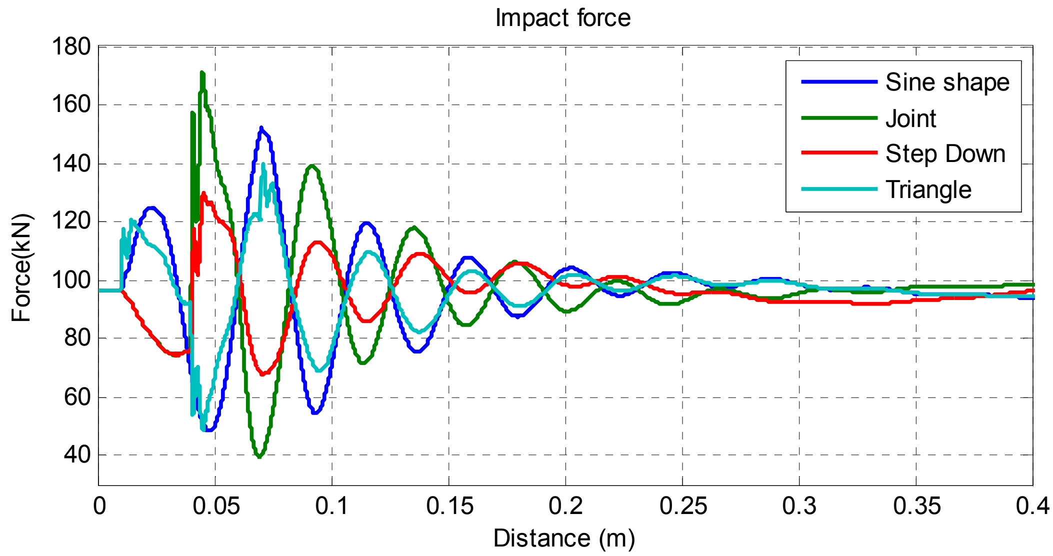


**Figure 12** – vertical wheel/rail force induced by the 4 simulated defects

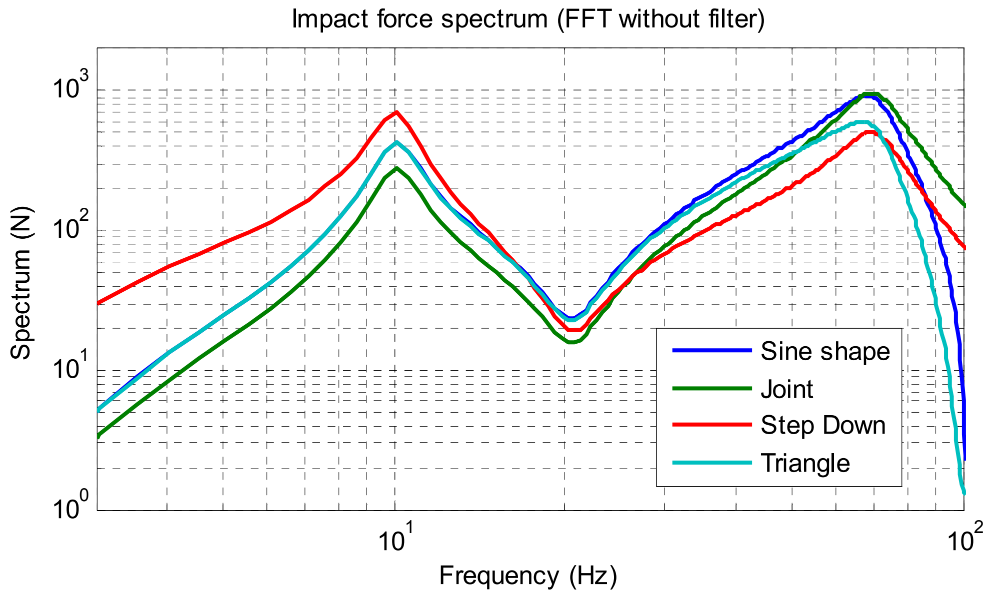


**Figure 13** – spectra of vertical wheel/rail force induced by the 4 simulated defects

Finally figure 14 shows a comparison of the vehicle receptance seen from the rail, reconstructed by means of formula (1)'' with the theoretical receptance directly computed from vehicle parameters.

Agreement between reconstructed and real receptance is perfect for all defects but the step-down one. The difference related to the step-down defect is related to a signal processing problem and should not be considered as relevant (rectangular windowing of a time signal not equal to 0 at one boundary of the window).

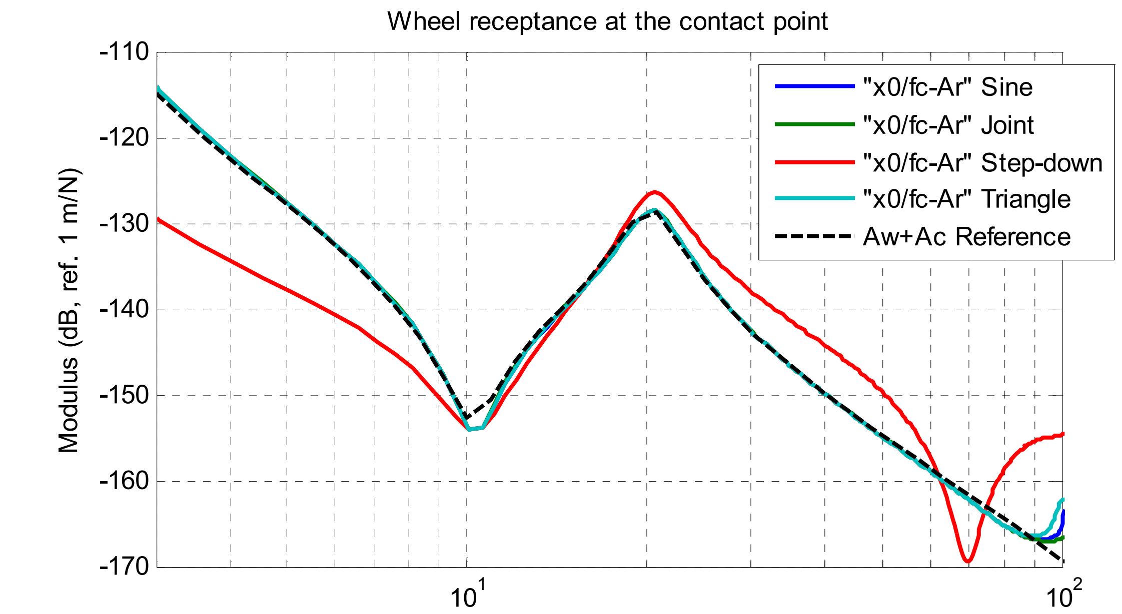


Figure 14 – **Comparison of reconstructed and of direct spectra of vehicle receptance**

### 4.5.3 Conclusion

These simulations are encouraging to promote the indirect method.

Reasonable defects with amplitudes less than 1 mm peak to peak, or rail joints with a reasonable width (3 cm in case of step down or 5 cm with no step down) lead to high dynamic forces for low speeds of about 10 kph.

At such speeds, the dynamic force component related to wheel and track current unevenness is expected to be much lower than that delivered by the calibrated defect, assuming good wheel and rail rolling surfaces. In case of doubts, wheels can be re-profiled and rail sections in the +/- 1 meter surrounding the measurement section can be ground before the measurements.

# 5. SIMULATING THE PERFORMANCE OF THE VEHICLE

The following section will describe a procedure for simulating the performance of a new vehicle in terms of its ground vibration emissions. The procedure is primarily based on simulation tools currently available but additional functionality expected from future development is also discussed. Important aspects when modelling the vehicle are described as well as relevant test cases.

## 5.1 REQUIREMENTS ON SIMULATION TOOL

As already described in this document the problem with ground vibration from trains need to be addressed with a system approach considering the vehicle, the track and the ground as three crucial sub-systems. Hence a prediction tool for evaluating the performance of a new vehicle design needs to include models of all sub-systems and be able to handle the interaction between them. Traditionally the vehicle manufacturer is asked to provide the customer with predictions of the vehicle performance prior to purchase. After manufacturing, the vehicle undergoes type tests which hopefully show a performance in line with the predicted one. In the case of ground vibration it is desired to do a type test of the vehicle but avoid the use of test tracks for the reasons described under section 4.2. In other words a type test should only be done for one of the tree sub-systems. This is only possible if the prediction tool enables replacing the vehicle model by a measured vehicle influence. This can be done if the vehicle and the track are modelled with their respective receptances (displacement over force). One example of such a tool is the TRAFFIC software developed by the department of civil engineering at the Katholieke Universiteit Leuven (KUL). TRAFFIC (http://bwk.kuleuven.be/bwm/traffic ) is the tool that has been used for the prediction of vehicle influence within RIVAS and the following procedure is written to be used with TRAFFIC or a tool with a similar modelling approach.

The influence of parametric excitation is currently not accounted for in TRAFFIC or any other tool known to the author. Considering the large contribution this excitation may give to the vibration level it needs to be included in a tool used for virtual testing of new vehicles. Simulations of the wheel-rail contact forces from parametric excitation have been done using a finite element, time domain model of the track. This functionality should be included in a future tool for virtual testing however without compromising a simulation approach which enables the replacement of a simulated vehicle influence by a measured vehicle influence.

### 5.1.1 The vehicle model

The complexity of the vehicle model will determine the level of detail to which the vehicle influence can be predicted. The unsprung mass, the wheel OOR and to some extend the primary suspension are the most important vehicle parameters to consider [1]. The most basic model would therefore include only these parameters which in many cases would be sufficient to give an adequate prediction. However when considering numerous combinations of vehicle, track and soil dynamics including extreme cases there is a possibility that other vehicle parameters such as the secondary suspension or the distance between bogies and axles can have an influence on the vibration response. On the other hand a comprehensive model such as the ones used for predicting the running dynamics of the vehicle is not necessary. These include elements for suspension in vertical, lateral and longitudinal direction as well as the suspension of individual components such as motor and gearbox. Trials with these kinds of models have shown that no additional information relevant for the excitation of ground vibration is given compared to using a model that includes only the vertical dynamics of the vehicle. The simplified model which has proven to be sufficient includes the primary and secondary suspension together with the carbody mass, bogie frame mass and the unsprung mass, see figure 15.

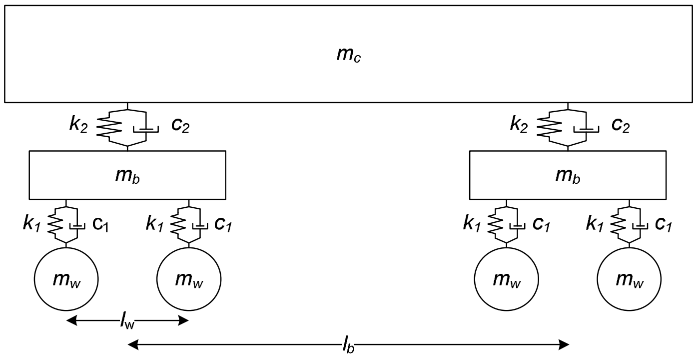


**Figure 15. Multi-body model for the vertical dynamic behaviour of the vehicle, including the car body, bogie frame and unsprung mass, $m_c$, $m_b$, $m_w$; the primary and secondary suspension stiffness, $k_1$, $k_2$ and damping, $c_1$, $c_2$. The parameters lw and lb gives the distance between axles and bogies respectively.**

In the TRAFFIC software the vertical point and cross receptances of the four wheels are calculated as a pre-processing and are used as input to the ground vibration simulation. Figure 3 illustrates the concept of receptance as displacement over force.

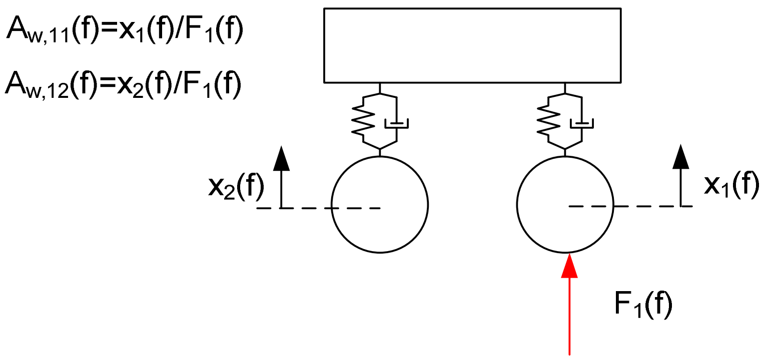


**Figure 16. . Point receptance $A_{w,11}$ and cross receptance $A_{w,12}$ for one bogie.**

The point receptance $A_{w,11}$ gives the displacement of wheel 1 as a result of the excitation of wheel 1 while the cross receptance $A_{w,12}$ gives the displacement of wheel 2 resulting from excitation of wheel 1. The frequency dependent receptances are assumed to be reciprocal in the linear model meaning that $A_{w,12}=A_{w,21}$. The complete model in figure 16 can therefore be described by 10 unique receptances: $A_{w,11}$, $A_{w,12}$, $A_{w,13}$, $A_{w,14}$, $A_{w,22}$, $A_{w,23}$, $A_{w,24}$, $A_{w,33}$, $A_{w,34}$, $A_{w,44}$. Figure 17 presents one example of the point receptance calculated for one wheel of the model. The receptance has been calculated for three different values of the unsprung wheelset mass to illustrate the influence from this parameter on the frequency dependent receptance.

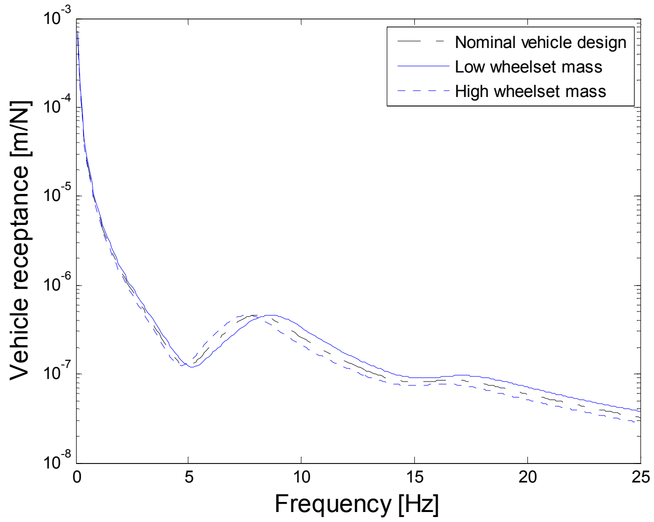


**Figure 17. Wheelset point receptance calculated with the vehicle model in figure 15. Three different values of the unsprung wheel set mass have been used to illustrate the influence on the receptance [1]**

When the vehicle has been manufactured the point and cross receptances are measured, either by the direct method, exciting the wheelsets and measuring the resulting displacement, or by the indirect method, rolling the vehicle over the test track with the artificial defect. Both methods result in the point and cross receptance. A new simulation is then done with the same track and ground parameters as for the first prediction but with the calculated vehicle receptances replaced by the measured receptances.

### 5.1.2 The track model

The track model in TRAFFIC includes the rail, the railpad, the sleeper and a ballast layer. The model is a 2.5D representation of a track which is invariant in the longitudinal direction. This means that the track stiffness is constant along the track and hence the excitation caused by discrete sleeper support is not included. A future tool for type testing new vehicles should include also this part of the excitation considering that it in many cases gives a significant contribution to the vibration level. The track model could either be a generic model of e.g. a standard UIC 60 track or it could be given parameter values representative for the track on the customer's network. If provided by the customer the track model could even be replaced by a measured track receptance.

### 5.1.3 The ground model

As for the track model the ground model can either be chosen to represent a generic case or a specific site. This choice might depend on if the vehicle will operate on a large network with many different types of soil supporting the track or on a small network where problematic sites are easier to identify. A future standard for virtual testing could include a number of reference ground models representative for different conditions in Europe as well as requirements on how a proper soil characterization is done if the customer decides to choose this option.

## 5.2 Test cases

The test cases should be defined in the customer's requirement specification of the new vehicle. These could be generic based on published measurement protocols e.g. AEA [9] or the one defined within RIVAS [10]. The test cases could also be tailored to reflect certain issues that the customer is experiencing on its particular network. E.g. the virtual measurement positions could be placed at a distance from the track where the most nearby buildings are located on the network. Another example is the selection of speed which together with the sleeper distance will determine the frequency of the parametric excitation. Here follows a few remarks important to consider when defining the test cases.

### 5.2.1 Virtual measurement positions

The first aspect to consider when choosing the measurement positions is the separate contributions from quasi-static and dynamic excitation. By placing the sensors close to the track the quasi-static contribution will be large and both the frequency content and level of the vibration might differ substantially compared to the vibration a few meters further away from the track. The quasi-static excitation is governed by the static axle loads and the speed of the vehicle. The influence from parametric excitation and vehicle resonances excited by wheel and rail irregularities will be less visible in a vibration spectrum measured close to the track. The distance from the track where the dynamic excitation will become dominant will differ from one site to another however as a rule of thumb sensors should be positioned not closer than 8 meters from the track center line to ensure that the dynamic excitation is dominating the response. One benefit of using virtual testing is that a lot of sensors can be used and that the quasi-static and dynamic excitations can be studied separately. This allows for a more robust evaluation with sensors at several distances from the track and in more than one track cross-section.

### 5.2.2 Vehicle speed

The speed of the vehicle will influence both the level and the frequency content of the excitation. A high speed lead to a large excitation and it might therefore be in the customer's interest to evaluate the performance at the top speed of the vehicle. On the other hand in a resonant system the maximum response is not necessarily achieved at maximum excitation and a lower vehicle speed could be a more critical test case if it e.g. leads to a parametric excitation which coincides with a poorly damped resonance in the vehicle-track-ground system. Again the virtual approach offers the possibility to test at several different speeds without imposing huge cost or work load.

### 5.2.3 Summary

From the above we conclude that it is feasible to use the measurement method with the fabricated rail defect to assess the vehicle's receptance with sufficient accuracy and as a next step to apply this measured receptance to predict the vehicle's performance in a wide range of systems consisting of different track types and qualities, soil characteristics and even vehicle speeds. This makes this method a very cost efficient approach to demonstrate the vehicle's quality and to optimize the vehicle for a wide range of operational applications.

# 6. PERFORMANCE MONITORING OF IN-SERVICE VEHICLES

When carrying out measurements of ground vibrations at some distance from the track and under mixed traffic conditions, it is usually noticed that one train causes higher vibration levels than the other. Typically, freight trains stand out for their generating high vibration levels. Reasons for that can be listed as follows:

- Wheel defects such as wheel flats and polygonisation may cause strong excitation of the track, particularly at relatively low speeds.
- Malfunctioning of the suspension (it is reported from field observations that the suspension is often short circuited due to corrosion) which causes the whole car body to act as the unsprung mass
- Freight trains generally include a locomotive, with heavy powered bogies (up to 1500 kg per bogie), which at close distance from the track causes high vibration levels if the unsprung mass is of similar weight as the entire bogie.

High dynamic forces acting on the track may cause increased track deterioration. For that reason, in some networks the dynamic and quasi-static forces on the track executed by the passing vehicles is monitored, most often by strain gauges attached to the rail. In combination with vehicle recognition systems, the monitoring results can feed into wheel and wagon maintenance schemes. These are sometimes encouraged by raising a penalty fee from the wagon owner or train operator running the particular wagon(s).

Wagons that cause high dynamic forces are well capable of generating high ground vibration levels. From today's practice we can conclude that there is no need for further specification of the track on which these vehicles run. The risk that a vehicle is identified as malfunctioning due to a coincidental resonance in the interaction between wagon and track (so that part of the resonance occurring could be attributed to the track) is almost negligible. However, if at some time in future there would be a need to monitor the vibration performance of in service vehicles with considerably better performance than the average freight wagon, then the track influence would need to be considered.

In that case, what would the track at the monitoring position have to look like? And what would be the measurement set up? Preferably the track would need to be as stiff and heavy as possible. One could consider a limited stretch of track with heavy sleepers or reduced sleeper distance, in combination with stiff rail pads in order to achieve this. The measurement set up would consist of a range of accelerometers, mounted on the foot of the rail.

The railway operators present in the RIVAS project have pointed out that the chance that they would be prepared to construct such a stretch of track, only for the purpose of monitoring the vibration performance of in service vehicles, is negligible. Currently, that is due to the fact that there is no legislation and no legal limits for ground vibration have to be respected. However, when such legislation would enter into force, it would be advisable to reconsider this position, as it is far more cost efficient to maintain the vehicles in a good condition than to install mitigation measures into the existing track or into the ground.

# 7. CONCLUSIONS

The current report describes the work done in Task 1.4 of Work package 1 of the RIVAS project. The work carried out in this task leads to the conclusion that it is not feasible to define a standard track and soil quality for compliance testing of new vehicles, even though it was the original intention of this task to define such a standard track, much like the EN3095 track used for compliance testing according to the TSI Noise.

As a next step, it was considered to define a standard European virtual reference track with good vibration performance (i.e. a fairly stiff track with very low unevenness) on a reference soil. If one would know all the relevant parameters of such a reference track and soil, one could collect the parameters of the new vehicle under concern and transfer these to the virtual reference track and soil in order to predict the overall system performance. This approach however would not be feasible because of the wide spread of different track and soil types in Europe. RIVAS measurement results on a range of different sites quite clearly show that there is no such thing as a standard European soil.

What remains is a procedure where vehicle manufacturer and vehicle purchaser agree on a set of typical track and soil types and that the vehicle is virtually tested against each of these sets. This is a cost efficient method in principle, but it still requires the main parameters of the vehicle to be assessed in practice. For this assessment, two methods have been developed in Task 1.4 and these methods have been tested using mathematical models.

The first method is a stationary method, to be applied for a vehicle in the workshop, and involves assessment of the receptance of a single wheel (and suspension), exciting the wheel with a shaker. This is a fairly simple, straightforward method although it is elaborate and time consuming if different wheel and axle configurations occur on one vehicle.

The second method is a pass-by method, where the vehicle rolls over a prefabricated artificial rail defect. The modelling shows, that a fairly small defect introduces sufficient vibrational energy into the track and wheel, in order to be able to assess the vehicle's receptance accurately. The train can run over the defect with fairly low speed. If the method is developed further, manufacturers could agree on a standard defect and track build up in order to achieve comparable results.

It is recommended that both the stationary and the pass-by method be applied in practice in order to acquire some experience. A task force under UNIFE or UIC could then further develop and standardise the method into a useful tool for vehicle optimisation.

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